NASA TECHNICAL NOTE

NASA TN D-6075
C. 1

NASA TN D-6075

ANALYSIS AND COMPUTER PROGRAM FOR EVALUATION OF ROTATING INCOMPRESSIBLY LUBRICATED PRESSURIZED THRUST BEARINGS

by David P. Fleming

Lewis Research Center

Cleveland, Ohio 44135

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • FEBRUARY 1971

1. Report No. NASA TN D-6075	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle ANALYSIS AND COMPUTER PROGRAM FOR EVALUATION OF ROTATING INCOMPRESSIBLY LUBRICATED PRESSURIZED THRUST BEARINGS		5. Report Date February 1971
		6. Performing Organization Code
7. Author(s) David P. Fleming		8. Performing Organization Report No. E-5938
		10. Work Unit No. 129-03
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Note
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D. C. 20546		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

An analysis and FORTRAN IV computer program are presented to enable rapid evaluation of pressurized thrust bearing designs using an incompressible lubricant. Included in the analysis are the effects of two self-acting journal bearings which may be used to provide a radial load capacity. Bearing load, torque, lubricant flow rate, and other quantities of interest are calculated. Either orifice or capillary restrictors may be used and effects of bearing rotation are included. Program input and output can be in U. S. Customary or metric (SI) units. Analytical predictions agreed well with experimental data from a series-hybrid fluid-film rolling-element bearing.

17. Key Words (Suggested by Author(s))		18. Distribution Statement
Bearings	Hydrostatic bearings	Unclassified - unlimited
Externally pressurized bearings	Thrust bearings	

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	28	$3.00

*For sale by the National Technical Information Service, Springfield, Virginia 22151

ANALYSIS AND COMPUTER PROGRAM FOR EVALUATION OF ROTATING INCOMPRESSIBLY LUBRICATED PRESSURIZED THRUST BEARINGS

by David P. Fleming

Lewis Research Center

SUMMARY

An analysis and FORTRAN IV computer program are presented to enable rapid evaluation of pressurized thrust bearing designs using an incompressible lubricant. Included in the analysis are the effects of two self-acting journal bearings which may be used to provide a radial load capacity. Bearing load, torque, lubricant flow rate, and other quantities of interest are calculated. Either orifice or capillary restrictors may be used and effects of bearing rotation are included. Program input and output can be in U. S. Customary or metric (SI) units. Analytical predictions agreed well with experimental data from a series-hybrid fluid-film rolling-element bearing.

INTRODUCTION

For the evaluation of a series-hybrid fluid-film ball bearing (ref. 1), the investigators used, for the fluid-film bearing, a pressurized thrust bearing in conjunction with a self-acting journal bearing. The lubricant was the same as that used for the ball bearing, namely, a type II ester oil. In the series-hybrid bearing (fig. 1), a fluid-film bearing and a ball bearing are coupled in series. Each component bearing carries the full system load, but the two bearings share the speed. One element of the fluid-film bearing rotates at shaft speed. The other element of the fluid-film bearing rotates with the inner race of the ball bearing at a speed less than shaft speed. The ball bearing outer race is stationary. The intermediate member rotates at a speed such that the torques of the fluid-film and ball bearings are equal. As shown in reference 1, the series-hybrid bearing has the potential of substantially increasing rolling-element bearing fatigue life at high speed.

Since the speed sharing between the fluid-film and ball bearings depends on the torque characteristics of the two component bearings, the bearing designer must properly size the fluid-film bearing to get a useful reduction in ball bearing speed and provide adequate load capacity.

Self-acting journal bearings have been quite thoroughly investigated (see, for example, refs. 2 and 3). Surprisingly, there appears to be no published information on rotating, compensated, pressurized thrust bearings using incompressible lubricants. Reference 4 gives design information for hydrostatic bearings with relative motion between the bearing parts. With liquid lubricants, however, centrifugal effects due to bearing rotation can significantly alter the bearing's characteristics. Formulas have been given for the load capacity of rotating uncompensated pressurized thrust bearings (for example, ref. 2, p. 203). The disadvantage of this type of bearing is that it has no stiffness and therefore cannot accept varying loads. It is, fortunately, a straightforward matter to add the effects of compensating restrictors (orifices or capillaries) to the analysis of an uncompensated pressurized bearing.

The objectives of this report are the following:

(1) To analyze an orifice (or capillary) compensated thrust bearing, including the rotational effects and the effects of any journal bearings adjoining the thrust bearing

(2) To present a digital computer program to carry out the analysis

(3) To give sample results of the analysis, and to compare these results with experimental data from the series-hybrid bearing

SYMBOLS

C_d	orifice discharge coefficient
C_i	radial clearance in inner journal bearing, in. (m)
C_o	radial clearance in outer journal bearing, in. (m)
d	orifice or capillary diameter, in. (m)
h	thrust bearing clearance, in. (m)
L	journal bearing length, in. (m)
L_R	capillary length, in. (m)
n	number of restrictors
p	pressure, lb/in.^{2} (N/m^2)
Q	lubricant flow rate, in.^{3}/sec (m^3/sec)
R	radius, in. (m)
Re	film rotational Reynolds number, $R\lvert w_1 - w_2\rvert h\rho/\mu$
r	radial coordinate, in. (m)
T	bearing torque, in.-lb (N-m)
v	velocity, in./sec (m/sec)
W	bearing load, lb (N)

z	axial coordinate, in. (m)
ϵ	journal bearing eccentricity ratio
θ	circumferential coordinate, rad
μ	lubricant dynamic viscosity, lb-sec/in.^{2} (N-sec/m^2)
ρ	lubricant density, lb/in.^{3} (kg/m^3)
ω	angular velocity, rpm or rad/sec
ω_o	mean angular velocity, eq. (7), rpm or rad/sec

Subscripts:

c	restrictor exit
i	inner
j	journal bearing
min	minimum
o	outer
p	pocket
r	radial direction
s	supply
z	axial direction
θ	circumferential direction
1	upper thrust surface
2	lower thrust surface

ANALYSIS

The bearing to be analyzed appears in figure 2. It comprises a thrust bearing and two journal bearings at the inside and outside thrust bearing radii. The journal bearings enter into the analysis only as they contribute additional constant resistances to the throughflow of lubricant and as they add to the bearing torque. Their lengths may be set to zero when they are not present. A circle of orifices is at radius R_c. Alternatively, the bearing may have capillary restrictors at radius R_c. The number of orifices or capillaries is assumed large enough to constitute a line source of lubricant. The thrust face clearance varies with radius in a stepwise manner as shown. The lubricant is supplied to the orifices at pressure p_s; it leaves the bearing at the reference pressure $p = 0$.

The starting point of the analysis is the Navier-Stokes equations for incompressible flow (ref. 5). The following assumptions are then imposed:

(1) The flow is laminar and steady with time.

(2) Rotational symmetry prevails, that is, $\partial/\partial\theta = 0$.

(3) Pressure in the thrust bearing is constant in the axial direction.

(4) There are no body forces acting on the fluid. (Centrifugal force appears as an inertia term.)

(5) The axial velocity is negligibly small in the thrust bearing.

With these assumptions, the Navier-Stokes equations become, in cylindrical coordinates,

$$\rho v_r \frac{\partial v_r}{\partial r} - \rho \frac{v_\theta^2}{r} = -\frac{dp}{dr} + \mu \frac{\partial^2 v_r}{\partial z^2} \tag{1a}$$

$$\frac{\rho v_r}{r} \frac{\partial}{\partial r}(r v_\theta) = \mu \frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(r v_\theta)\right) + \mu \frac{\partial^2 v_\theta}{\partial z^2} \tag{1b}$$

The continuity equation has been incorporated into equation (1a). In a rotating bearing, the radial fluid velocity v_r is much less than the circumferential velocity v_θ. Thus first term of equation (1a) can be neglected relative to the second. It is assumed that viscous forces predominate over inertia forces. Thus the first term of equation (1b) can be eliminated. (These are standard assumptions in lubrication work. See, for example, ref. 3, p. 68). Equations (1a) and (1b) become

$$-\rho \frac{v_\theta^2}{r} = -\frac{dp}{dr} + \mu \frac{\partial^2 v_r}{\partial z^2} \tag{2a}$$

$$\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(r v_\theta)\right) + \frac{\partial^2 v_\theta}{\partial z^2} = 0 \tag{2b}$$

A solution of equation (2b) is

$$v_\theta = r\omega_2 - r(\omega_2 - \omega_1)\frac{z}{h} \tag{3}$$

where z is measured from the lower thrust surface (fig. 2). Equation (3) may be substituted into equation (2a) and the result integrated twice in z. After the boundary conditions $v_r = 0$ at $z = 0$ and $v_r = 0$ at $z = h$ are applied, the result is

$$v_r = \frac{1}{2\mu}\frac{dp}{dr}(z^2 - zh) - \frac{\rho r}{\mu}\left[\frac{\omega_2^2}{2}(z^2 - zh) - \frac{\omega_2(\omega_2 - \omega_1)}{3h}(z^3 - zh^2) + \frac{(\omega_2 - \omega_1)^2}{12h^3}(z^4 - zh^3)\right] \tag{4}$$

Lubricant Flow Rates

The quantity of lubricant flowing radially at any radius r is found from

$$Q = 2\pi r \int_0^h v_r dz \tag{5}$$

Substituting equation (4) into equation (5) and integrating yield

$$Q = \frac{\pi r h^3}{6\mu}\left(-\frac{dp}{dr} + \rho r \omega_o^2\right) \tag{6}$$

where ω_o is defined by

$$\omega_o^2 \equiv \omega_1\omega_2 + \frac{3}{10}(\omega_2 - \omega_1)^2 \tag{7}$$

The speed ω_o, which may be regarded as an average of ω_1 and ω_2, is used to calculate rotational effects. Equation (6) may now be integrated with respect to r to find the relation between flow and pressure. The clearance h has been assumed to vary in a stepwise manner. Thus, integration over any interval of constant clearance is straightforward. For example, integrating from $r = R_{po}$ to $r = R \le R_o$ gives

$$p_{po} - p = -\frac{6\mu Q_o}{\pi h_o^3}\ln\frac{R_{po}}{R} + \frac{\rho}{2}\left(R_{po}^2 - R^2\right)\omega_o^2 \tag{8}$$

Pressure drop through the journal bearings is easily calculated from the expression for flow in a narrow slot (ref. 3, p. 99). For the outer journal bearing, neglecting the effect of eccentricity,

$$p_o - p_{ref} = \frac{6\pi Q_o L_o}{\pi R_o C_o^3} \tag{9}$$

With this, total lubricant flow through the bearing can now be given as that flowing inward from the orifices and that flowing outward:

$$Q = Q_o - Q_i = \frac{\pi}{6\mu} \frac{p_c - \frac{\rho}{2}\left(R_c^2 - R_o^2\right)\omega_o^2}{\frac{L_o}{R_o C_o^3} + \frac{1}{h_p^3} \ln \frac{R_{po}}{R_c} + \frac{1}{h_o^3} \ln \frac{R_o}{R_{po}}} + \frac{\pi}{6\mu} \frac{p_c - \frac{\rho}{2}\left(R_c^2 - R_i^2\right)\omega_o^2}{\frac{L_i}{R_i C_i^3} + \frac{1}{h_p^3} \ln \frac{R_c}{R_{pi}} + \frac{1}{h_i^3} \ln \frac{R_{pi}}{R_i}} \tag{10}$$

The negative of Q_i is taken because, at any radius, outward flow is defined as positive.

The total flow Q must now be matched to the flow through the restıictors. For orifice restrictors, this is (ref. 4, p. 103)

$$Q = C_d n\pi \frac{d^2}{4} \sqrt{\frac{2(p_s - p_c)}{\rho}} \tag{11a}$$

and for capillary restrictors,

$$Q = \frac{n\pi d^4 (p_s - p_c)}{128 \mu L_R} \tag{11b}$$

Equation (11a) or (11b), as appropriate, may be combined with equation (10), and the pressure p_c downstream of the restrictors solved for algebraically. The flow rate Q can then be found from equation (10).

Bearing Loads

The thrust bearing load is given by

$$W = 2\pi \int_{R_i}^{R_o} pr\, dr \tag{12}$$

The pressure p is given by equation (8) for $R_{po} \le r \le R_o$, and by similar expressions for other radii. These expressions may be substituted into equation (12) to yield the load. For example,

$$W_{po} = 2\pi \int_{R_c}^{R_{po}} pr\, dr = \pi p_c \left(R_{po}^2 - R_c^2\right) - \frac{6\mu Q_o}{h_p^3}\left(R_{po}^2 \ln \frac{R_{po}}{R_c} - \frac{R_{po}^2 - R_o^2}{2}\right) + \pi \rho \omega_o^2 \left(\frac{R_{po}^2 - R_c^2}{2}\right)^2 \tag{13}$$

Inner and outer journal bearings (fig. 2) may be used to provide a radial load capacity. These are assumed to be purely self-acting bearings. For small length-to-diameter ratios, the load is adequately given by the short bearing approximation. From reference 3, (p. 84), for the inner bearing,

$$W_{ji} = \frac{\mu R_i \left|\omega_1 - \omega_2\right| L_i^3}{4C_i^2} \frac{\epsilon_i}{\left(1 - \epsilon_i^2\right)^2} \left[\pi^2\left(1 - \epsilon_i^2\right) + 16\epsilon_i^2\right]^{1/2} \tag{14}$$

where ϵ_i is the journal-bearing eccentricity ratio.

Bearing Torque

The velocity v_θ varies linearly across the film (eq. (3)). Thus, thrust bearing torque is easily calculated. For the annulus from $r = R_c$ to $r = R_{po}$,

$$T_{po} = \frac{\pi\mu \left|\omega_1 - \omega_2\right| \left(R_{po}^4 - R_c^4\right)}{2h_p} \tag{15}$$

Similar expressions apply to the other sections. For the inner journal bearing, neglecting the effect of eccentricity on torque, the torque is

$$T_{ji} = \frac{2\pi R_i^3 L_i \mu |\omega_1 - \omega_2|}{C_i} \tag{16}$$

A similar expression applies to the outer journal bearing. Total torque is merely the sum of the various component torques:

$$T = T_o + T_{po} + T_{pi} + T_i + T_{jo} + T_{ji} \tag{17}$$

COMPUTER PROGRAM

Though the preceding analysis is simple and straightforward, a considerable effort would be needed to calculate the large number of numerical results needed in evaluating several bearing designs and operating conditions. To reduce this effort, a computer program was written. The program is written in FORTRAN IV for the IBM 7094-II computers at Lewis Research Center. It can be used on any digital computer accepting FORTRAN IV programs. The program listing appears in the appendix. Figure 3 is a flow chart of the program. Four options are available in the program:

(1) Input and output may be in either U. S. Customary or metric (SI) units. To use metric units, set METRIC = .TRUE. in the input list.

(2) Orifice or capillary restrictors may be used. If the capillary length LR is not zero, capillary restrictors are assumed; if LR = O, orifice restrictors are assumed.

(3) Output may be for a specified set of clearances or a specified set of bearing torques. If TMAXX is zero, output will be for NH outer thrust face clearances (h_o) starting at h_o = HOO and proceeding up by increments of DH. If TMAXX is nonzero, the program will still start with h_o = HOO. It will then increase h_o by increments of DH until the torque decreases to TMAXX. (HOO must be chosen so the corresponding torque is greater than TMAXX). It will then make calculations for clearances such that bearing torque drops by increments of DT from TMAXX to TMIN. Under either option, the program stops incrementing clearance or torque whenever the bearing load becomes negative.

(4) Supply pressure PS may be specified. Or, if PS is not specified, the program will calculate a supply pressure based on the pressure available from centrifugal force at the radius of the restrictors:

$$p_s = \frac{1}{2} \rho R_c^2 \omega_1^2 \tag{18}$$

To have the program calculate p_s, set CENT = .TRUE.

As a check on whether the assumed laminar conditions prevail, the program calculates Reynolds numbers based on the thrust bearing film thickness and rotational speed:

$$Re = R|\omega_1 - \omega_2| h\rho/\mu \tag{19}$$

The three Reynolds numbers calculated are for the outer thrust face land (using R_o and h_o), for the inner thrust face land (using R_{pi} and h_i), and for the pocket (using R_{po} and h_p). If any one of these Reynolds numbers exceeds 1000, the flow is probably turbulent in that region (ref. 6). Computed results will then be inaccurate.

Program Input

All input is in NAMELIST format using the NAMELIST name BRG. Sample input appears in figure 4. Note that all data cards begin in column 2. Following is a description of the input variables:

FORTRAN name	Analysis symbol	Description
CD	C_d	orifice discharge coefficient
CENT		if .TRUE., PS calculated from eq. (18)
CI	C_i	radial clearance in inner journal bearing, in. (m)
CO	C_o	radial clearance in outer journal bearing, in. (m)
D	d	diameter of orifices or capillaries, in. (m)
DH		increment in thrust bearing clearance, in. (m)
DT		increment in bearing torque, in.-lb (N-m)
DW2		increment in speed ω_2, rpm
HII		initial value of inner thrust clearance h_i, in. (m)
HOO		initial value of outer thrust clearance h_o, in. (m)
HPP		initial value of pocket thrust clearance h_p, in. (m)
LI	L_i	length of inner journal bearing, in. (m) (may be zero)

FORTRAN name	Analysis symbol	Description
LO	L_o	length of outer journal bearing, in. (m) (may be zero)
LR	L_R	length of capillary restrictors, in. (m) (zero for orifice restrictors)
METRIC		if .TRUE., input and output are in metric (SI) units
MU	μ	lubricant dynamic viscosity, lb-sec/in.^{2} (N-sec/m^2)
N	n	number of orifice or capillary restrictors
NH		number of values of thrust clearance
NW2		number of values of ω_2
PS	p_s	lubricant supply pressure, lb/in.^{2} (N/m^2) (needed only if CENT = .FALSE.)
RC	R_c	radius of restrictor circle, in. (m)
RHO	ρ	lubricant density, lb/ft^3 (kg/m^3)
RI	R_i	inner radius of bearing, in. (m)
RO	R_o	outer radius of bearing, in. (m)
RPI	R_{pi}	inner radius of pocket, in. (m)
RPO	R_{po}	outer radius of pocket, in. (m)
TMAXX		maximum bearing torque for which calculations are desired, in.-lb (N-m)
TMIN		minimum bearing torque for which calculations are desired, in.-lb (N-m)
W1	ω_1	speed of upper thrust face, rpm
W22		initial value of ω_2, rpm

Any number of cases may be run at one time.

Program Output

Figure 5 illustrates program output. All input data are printed at the beginning of each case. After the input data, the speed of the upper thrust face is printed, followed by the supply pressure if that was calculated by the program (CENT = .TRUE.). The following line shows the speeds of the lower and upper thrust faces, ω_2 and ω_1, and the calculated mean speed ω_o (eq. (7)). The next line gives the torque due to the journal bearings and the loads the journal bearings will carry at an eccentricity ratio of 0.5.

The following line comprises column headings. These headings include (1) clearances h_o, h_i, and h_p, (2) total bearing load W and torque T, (3) total lubricant flow Q, (4) load contributions of outer and inner thrust face lands, W_o and W_i, (5) load contribution of the pocket for $r > R_c$ and $r < R_c$, (6) torque contributions of the outer and inner thrust face lands and pocket, (7) lubricant flow rates outward and inward from orifice radius R_c (inward flow is negative), (8) pressure p_c downstream of the restrictors and pressures p_{pi} and p_i at radii R_{pi} and R_i, (9) pressures p_{po} and p_o at radii R_{po} and R_o, (10) film rotational Reynolds numbers for outer and inner thrust face lands and pocket, and (11) radius $R_{p_{min}}$ and pressure p_{min} and the derivative $dp/dr\big|_{r=R_o}$. If $R_{p_{min}} < R_o$ or $dp/dr\big|_{r=R_o} > 0$ the pressure somewhere in the bearing film is subambient. Under this condition, a real bearing will cavitate; computer results are inaccurate. Pressure p_{min} is found by extrapolating from $r = R_o$ until $dp/dr = 0$ using equation (8).

SAMPLE RESULTS

The pressurized thrust bearing for which results are given is illustrated in figure 1 as part of a series-hybrid rolling-element bearing. This bearing was evaluated experimentally in reference 1. The centrifugal field in the rotating upper thrust member was used to pressurize the lubricant. Bearing dimensions are given in U. S. Customary Units in the computer program output of figure 5. Figure 6 shows bearing thrust load as a function of outer thrust face clearance h_o. Typical of externally pressurized bearings, the stiffness becomes very small as the clearance approaches zero. (Stiffness is given by the slope of the load-clearance curve.) When the lower thrust face rotates at the same speed as the upper face, the load at zero clearance is somewhat higher than when the lower thrust face is stationary. At larger clearances, however, the load is less when both faces rotate. Centrifugal force is responsible for this difference. The rotative speed used to calculate centrifugal effects (ω_o) is between the upper and lower thrust face speeds (eq. (7)). Thus, with both faces rotating, centrifugal effects are greater than with only one rotating.

Figure 7 gives the variation of lubricant throughflow with change in clearance. As expected, flow increases with increasing clearance. Increased centrifugal effects increase the slope of the curve when both thrust faces rotate.

Figure 8 shows the torque of the bearing assembly as a function of clearance. Torque decreases with increasing clearance. When the lower thrust face rotates at half the upper thrust face speed, torque is halved compared to the case when $\omega_2 = 0$, since torque is directly proportional to the speed difference between the two faces. When the two faces rotate at the same speed, the torque is, of course, zero.

Figure 9 shows the variation of load with lower thrust face speed for various bearing torques. This has been plotted from the alternate form of program output, in which clearance is varied to give preassigned torque values. This form is convenient for determining bearing speed for given load and torque, as was necessary in reference 1.

In figures 6 and 9 the curves stop before reaching zero load. This is because, as the bearing clearance increases, cavitation eventually sets in, making load calculations inaccurate. The presence of cavitation is indicated by negative intermediate pressures in the program output.

Comparison with Experiment

Figure 10 compares speed sharing predictions of the analysis with experimental data from reference 1. Oil viscosities corresponding to experimentally measured ball-bearing outer-race temperatures were used in the computer program. Points for the analytical curve were obtained from plots similar to figure 9 using measured bearing torque.

The experimental apparatus had a centrifugal oil supply. Therefore, supply pressure varied with shaft speed according to equation (18). When the supply pressure becomes high enough (high enough shaft speed) to lift off the fluid-film thrust bearing, the intermediate speed drops abruptly, according to the analysis. After lift-off, intermediate speed rises at a slightly lower rate than shaft speed.

Agreement is generally good between analysis and experiment, though experimental lift-off speeds and intermediate speeds are somewhat higher than predicted. Possible reasons for this are (1) the analysis assumes a line source of lubricant in the fluid-film thrust bearing, whereas the experimental bearing has only four orifices, and (2) the fluid-film bearing may be cooler than the ball bearing outer race, resulting in higher oil viscosities than assumed. Fluid turbulence probably did not affect the results significantly, since the highest Reynolds number predicted by the analysis is 1020 for the data of figure 10. This is only slightly above the threshold of 1000 reported in reference 6.

CONCLUDING REMARKS

An analysis has been presented and a computer program has been developed to enable rapid evaluation of rotating pressurized thrust bearing designs using an incompressible lubricant. Included in the analysis are the effects of two self-acting journal bearings which may be used to provide a radial load capacity. Bearing load, torque, lubricant flow rate, and other quantities of interest are calculated. Either orifice or capillary compensation can be used and effects of bearing rotation are included. Program input and output can be in either U.S. Customary or metric (SI) units. The computer program was written in FORTRAN IV; it can be used on most modern digital computers.

Analytical predictions agree well with experimental data from a series-hybrid fluid-film rolling-element bearing.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 12, 1970,
129-03.

APPENDIX - COMPUTER PROGRAM FOR ANALYSIS OF PRESSURIZED ROTATING THRUST BEARINGS

```
$IBFTC RCFF2
      REAL LI,LO,LR,MU
      LOGICAL DELT,CENT,METRIC
      NAMELIST /BRG/ CD,CI,CO,D,DH,DT,DW2,HII,HOO,HPP,LI,LO,LR,MU,N,NH,
     1 NW2,PS,RC,RHO,RHOM,RI,RO,RPI,RPO,TMAXX,TMIN,W1,W22,CENT,METRIC
C  CD = ORIFICE DISCHARGE COEFFICIENT    NH = NUMBER OF HO'S
C  CI = INNER JNL BRG RAD CLEARANCE      NW2= NUMBER OF W2'S
C  CO = OUTER JNL BRG RAD CLEARANCE      PS = SUPPLY PRESSURE, PSI (N/M2)
C  D  = ORIFICE OR CAPILLARY DIAMETER    RC = RADIUS OF ORIFICE CIRCLE
C  DH = INCREMENT IN HO,HI,AND HP        RHO= LUBRICANT DENSITY, LB/FT3 (KG/M3)
C  DT = INCREMENT IN TORQUE              RI = INSIDE BEARING RADIUS
C  DW2= INCREMENT IN W2, RPM             RO = OUTSIDE BEARING RADIUS
C  HII= INITIAL INNER THRUST CLEARANCE   RPI= INNER POCKET RADIUS
C  HOO= INITIAL OUTER THRUST CLEARANCE   RPO= OUTER POCKET RADIUS
C  HPP= INITIAL POCKET CLEARANCE         TMAXX= MAXIMUM TORQUE, IN LB (N M)
C  LI = INNER JOURNAL BEARING LENGTH     TMIN= MINIMUM TORQUE, IN LB (N M)
C  LO = OUTER JOURNAL BEARING LENGTH     W1 = SHAFT SPEED, RPM
C  LR = CAPILLARY LENGTH                 W22= INITIAL LOWER THRUST FACE SPEED,RPM
C  MU = LUBRICANT DYNAMIC VISCOSITY      CENT  IF.TRUE. PS CALC FROM CENT FORCE
C  N = NUMBER OF ORIFICES                METRIC  IF.TRUE. USE METRIC UNITS
C  ALL LENGTHS ARE INCHES (METERS)
      F(E) = E/4./(1.-E*E)**2*SQRT(PI*PI*(1.-E*E)+16.*E*E)
C  F(E) IS FACTOR TO COMPUTE JOURNAL BRG LOAD (FROM SHORT BRG THEORY)
      DATA PI/3.14159265/,CD/.6/,CO,CI/1.,1./,HII,HOO,HPP/0.,0.,.1/
      DATA DH/.0001/,LI,LO,LR/0.,0.,0./,W22/0./,METRIC/.FALSE./
      DATA TMAXX,DT,TMIN/0.,0.,0./
      ICMAX = 5
C  ICMAX = MAXIMUM NUMBER OF ITERATIONS TO GET TORQUE WANTED
5     READ (5,BRG)
      DHP = HPP - HOO
      DHI = HII- HOO
      EPST = .01
      IF (LR.EQ.0.) WRITE (6,6)
      IF (LR.NE.0.) WRITE (6,7)
6     FORMAT (49H1  ANALYSIS OF ORIFICE COMPENSATED THRUST BEARING )
7     FORMAT (51H1  ANALYSIS OF CAPILLARY COMPENSATED THRUST BEARING )
      RHOM = RHO
      IF (.NOT.METRIC) RHOM = RHO/1728./386.
      RC2 = RC*RC
      RO2 = RO*RO
      RI2 = RI*RI
      RPI2 = RPI*RPI
      RPO2 = RPO*RPO
      W1R = W1*PI/30.
      IF (CENT) PS = RHOM*RC2*W1R*W1R/2.
      WRITE (6,BRG)
      IF (CENT.AND..NOT.METRIC) WRITE (6,10) W1,PS,RC
      IF (CENT.AND.METRIC)      WRITE (6,11) W1,PS,RC
      IF (.NOT.CENT) WRITE (6,12) W1
10    FORMAT (5HK  W1 F10.1,5H RPM, 4X 27HCENTRIFUGAL SUPPLY PRESSURE
     1 G10.3,11HPSI AT RC = G10.3,6HINCHES)
11    FORMAT (5HK  W1 F10.1,5H RPM, 4X 27HCENTRIFUGAL SUPPLY PRESSURE
     1 G10.3,13H N/M2 AT RC = G10.3,7H METERS )
12    FORMAT (5HK  W1 F10.1,4H RPM )
      RKO = CD*FLOAT(N)*PI*D*D/4.*SQRT(2./RHOM)
```

```
      SKU = RKU*KKU
      XC = 128.*MU*LR/FLOAT(N)/PI/D**4
      SIX = 6.*MU/PI
      ROPL = ALOG(RO/RPO)
      RPOCL = ALOG(RPO/RC)
      RCPIL = ALOG(RC/RPI)
      RPIL = ALOG(RPI/RI)
C   FACTORS FOR COMPUTING THRUST BEARING LOADS AND LUBRICANT FLOWS
      PF2 = RHOM/2.*(RO2 - RPO2 )
      PF3 = RHOM/2.*(RPO2 - RC2 )
      PF4 = RHOM/2.*(RPI2 - RC2)
      PF5 = RHOM/2.*(RI2 - RPI2)
      WOF = SIX*(RO2*ROPL- (RO2-RPO2)/2.)
      WPOF = SIX*(RPO2*RPOCL - (RCP2-RC2)/2.)
      WPIF= SIX*((RC2-RPI2)/2. - RPI2*RCPIL)
      WIF = SIX*((RPI2-RI2)/2. - RI2*RPIL)
      W2 = W22
      E = .5
C   FACTORS FOR COMPUTING JOURNAL BEARING TORQUES,LOADS,AND FLOW RESISTANCES
      FE = F(E)
      TJOF = 2.*PI*MU*RO2*RO*LO/CO
      TJIF = 2.*PI*MU*RI2*RI/CI*LI
      WJIF = FE*MU*RI/CI/CI*LI**3
      WJOF = MU*RO*LO**3/CO/CO*FE
C   DO LOOP FOR RANGE OF SPEEDS
      DO 50 NW= 1,NW2
      HO = HOO
      HI = HII
      HP = HPP
      WO2 =W1*W2 + .3*(W1-W2)**2
      WO = SQRT (WO2)
      WRITE (6,15) W2,W1,WO
15    FORMAT (5HL  W2 F10.1,5H RPM, 3X 2HW1 F10.1,5H RPM, 3X 2HWO
     1 F10.1,4H RPM )
      WOR = WO2*PI*PI/900.
      W2R = W2*PI/30.
      DWR = W1R - W2R
      TJO = TJOF*DWR
      TJI = TJIF*DWR
      WJO = WJOF*DWR
      WJI = WJIF*DWR
      IF (.NOT.METRIC) WRITE (6,20)  TJO,TJI,E,WJO,WJI
      IF (METRIC)      WRITE (6,21)  TJO,TJI,E,WJO,WJI
20    FORMAT (42H   JOURNAL BEARING TORQUES TJO,TJI, IN LB  2G10.3,
     1 22H  LOADS AT ECC RATIO = F5.2,14H   WJO,WJI,  LB   2G10.3 )
21    FORMAT (40H   JOURNAL BEARING TORQUES TJO,TJI, N M  2G10.3,
     1 22H  LOADS AT ECC RATIO = F5.2,19H   WJO,WJI, NEWTONS   2G10.3 )
      IF (.NOT.METRIC) WRITE (6,25)
      IF (METRIC)      WRITE (6,26)
25    FORMAT ( 9HKHO,HI,HP 9H  LOAD,LB 2X 12HFLOW,IN3/SEC 2X 8HWO,WI,LB
     1 3X 10HWPO,WPI,LB 3X 8HTO,TI,TP 5X 5HQO,QI 5X 9HPC,PPI,PI 5X
     2 6HPPO,PO 2X 11HREYNOLDS NR 1X 12HR(PMIN),PMIN /
     3 3X 4HMILS 4X 12HTORQUE,IN LB 36X 5HIN LB 6X 7HIN3/SEC  7X 3HPSI
     4 9X 3HPSI 4X 11HREO,REI,REP 3X 8HDPOR(RO) )
26    FORMAT(11HK  HO,HI,HP 8H  LOAD,N 3X 12HFLOW, M3/SEC 3X 7HWO,WI,N
     1 4X 9HWPO,WPI,N 4X 8HTO,TI,TP 5X 5HQO,QI 5X 9HPC,PPI,PI 5X
     2 6HPPO,PO 2X 11HREYNOLDS NR 1X 12HR(PMIN),PMIN /
     3 6X 2HMM      8X 11HTORQUE, N M 36X 3HN M 7X 7H M3/SEC  6X 5HN/M2
     4 7X 5HN/M2 3X 11HREO,REI,REP 3X 8HDPOR(RO) )
```

```
      PR3 = PF3*WOR
      PR2 = PF2*WOR
      PR4 = PF4*WOR
      PR5 = PF5*WOR
      PRO = PR2 + PR3
      PRI = PR4 + PR5
      TTOF = PI*MU*DWR/2.
      TTPF = TTOF*(RPO2*RPO2 - RPI2*RPI2)
      TTIF = TTOF*(RPI2*RPI2 - RI2*RI2)
      TTOF = TTOF*(RO2*RO2 - RPO2*RPO2)
      REF = DWR*RHOM/MU
C   FACTOR FOR COMPUTING FILM ROTATIONAL REYNCLDS NUMBERS
      DELT = .FALSE.
      TMAX = TMAXX
C   DO LOOP FCR RANGE OF CLEARANCES
      DO 40 I=1,NH
      IC = 0
29    HO3 = HO**3
      HI3 = HI**3
      HP3 = HP**3
      HIM3 = AMIN1 (HI3,HP3)
      HOM3 = AMIN1(HO3,HP3)
      X1= SIX*LO*HOM3/RO/CO**3
      X2 = SIX*ROPL
      IF(HO3.NE.HOM3)  X2 = X2*HOM3/HO3
      X3 = SIX*RPOCL
      IF(HP3.NE.HOM3) X3 = X3*HOM3/HP3
      X4 = SIX*RCPIL
      IF(HP3.NE.HIM3) X4 = X4*HIM3/HP3
      X5 = SIX*RPIL
      IF(HI3.NE.HIM3)  X5 = X5*HIM3/HI3
      X6 = SIX *LI*HIM3/RI/CI**3
      XO = X1 + X2 + X3
      XI = X4 + X5 + X6
      HXI = HIM3/XI
      HXO = HOM3/XO
      HX = HXI + HXO
      IF (LR.EQ.0.) GO TO 293
CALCULATE PC FOR CAPILLARY COMPENSATION
      PC = (PS/XC - PRO*HXO - PRI*HXI)/(HX + 1./XC)
      Q = (PS - PC)/XC
      GO TO 295
293   PHX = (PRO*HXO + PRI*HXI)*HX
CALCULATE PC FOR ORIFICE COMPENSATION
      PC = (-PHX - SKO/2. + RKO*SQRT(PHX + SKO/4. + HX*HX*PS))/HX/HX
      Q = RKO*SQRT(PS-PC)
295   QO = (PC + PRO)/XO
      QI = -(PC + PRI)/XI
      PPI = PC + QI*X4 + PR4
      PIN= PPI + QI*X5 + PR5
      PPO= PC - QO*X3 + PR3
      PO = PPO - QO*X2 + PR2
      WO = -QO*WOF
      IF(HO3.NE.HOM3) WO = WO*HOM3/HO3
      WO = PI*(WO + (PPO + PR2/2.) *(RO2 - RPO2))
      WPO = -QO*WPOF
      IF(HP3.NE.HOM3) WPO= WPO*HOM3/HP3
      WPO = PI*(WPO+ (PC + PR3/2.) *(RPO2 - RC2))
      WPI = QI*WPIF
      IF(HP3.NE.HIM3)  WPI = WPI*HIM3/HP3
```

```
      WPI= PI*(WPI + (PC + PR4/2.) *(RC2-RPI2))
      WI = QI*WIF
      IF(HI3.NE.HIM3)  WI = WI*HIM3/HI3
      WI = PI*(WI + (PPI + PR5/2.) *(RPI2- RI2))
      FAC = 6.*MU*QO/PI
      IF(HO3.NE.HOM3)  FAC = FAC*HOM3/HO3
      QO = QO*HOM3
      QI = QI*HIM3
      TTO = 1.E6
      TTI = 1.E6
      TTP = 1.E6
      IF (HO.NE.O.) TTO = TTOF/HO
      IF (HI.NE.O.) TTI = TTIF/HI
      IF(HP.NE.C.) TTP = TTPF/HP
      T = TTO + TTP + TTI + TJO + TJI
      W = WO + WPO + WPI + WI
CALCULATE MAXIMUM FILM ROTATIONAL REYNOLDS NUMBERS FOR THRUST BEARING
      REO = REF*RO*HO
      REI = REF*RPI*HI
      REP = REF*RPO*HP
      DPDR = -FAC/RO + RHOM*RO*WOR
      RM2 = FAC/RHOM/WOR
      RM = SQRT(ABS(RM2))
      PMIN = PPO - FAC*ALOG(RM/RPO) + RHOM*WOR/2.*(RM2-RPO2)
      IF (RM2.LT.O.) RM = -RM
      IF (DELT.AND.ABS(T-TMAX).GT.EPST.AND.W.GE.O.) GO TO 35
      IF (.NOT.METRIC) WRITE (6,30) HO,W,Q,WO,WPO,TTO,QO,PC,PPO,REO,RM,
     2 HI,T,WI,WPI,TTI,QI,PPI,PO,REI,PMIN,   HP,TTP,PIN,REP,DPDR
      IF (METRIC)      WRITE (6,31) HO,W,Q,WO,WPO,TTO,QO,PC,PPO,REO,RM,
     2 HI,T,WI,WPI,TTI,QI,PPI,PO,REI,PMIN,   HP,TTP,PIN,REP,DPDR
30    FORMAT (/3PF8.3,OPG12.4,9G12.3/3PF11.2,OPG14.3, 9X 8G12.3/
     2 3PF11.2,47X OPG12.3,12X G12.3,12X 2G12.3)
31    FORMAT (/3PF10.5,OPG12.4,9G12.3/3PF13.4,OPG14.3,9X 8G12.3/
     2 3PF13.4,47X OPG12.3,12X G12.3,12X 2G12.3)
      IF (W.LT.O.) GO TO 50
35    HT = HO
      IF (ABS(T-TMAX).LE.EPST.AND.W1.NE.W2) GO TO 200
      IF (T.GT.TMAXX.AND..NOT.DELT.OR.W1.EQ.W2) GO TO 39
      IF (IC.GT.ICMAX) GO TO 200
      IC = IC + 1
      DELT = .TRUE.
10C   HO = HO + (TMAX-T)/(T1-T)*(H1-HO)
      IF (HO.LE.O.) HO = .005
      HI = HO + DHI
      HP = HO + DHP
      IF (ABS(T1-TMAX).LE.ABS(T-TMAX)) GO TO 29
      H1 = HT
      T1 = T
      GO TO 29
20C   TMAX = TMAX - DT
      IC = C
      DELT = .TRUE.
      IF (TMAX.EQ.O..OR.TMAX.LT.TMIN) GO TO 50
      GO TO 100
39    H1 = HO
      T1 = T
      HI = HI + DH
      HP = HP + DH
40    HO = HO + DH
```

```
50    W2 = W2 + DW2
      GO TO 5
      END
```

REFERENCES

1. Parker, Richard J.; Fleming, David P.; Anderson, William J.; and Coe, Harold H.: Experimental Evaluation of the Series-Hybrid Rolling-Element Bearing. NASA TN D-7011, 1970.
2. Pinkus, Oscar; and Sternlicht, Beno: Theory of Hydrodynamic Lubrication. McGraw-Hill Book Co., Inc., 1961.
3. Bisson, Edmond E.; and Anderson, William J.: Advanced Bearing Technology. NASA SP-38, 1964.
4. Rippel, Harry C.: Cast Bronze Hydrostatic Bearing Design Manual. Cast Bronze Bearings Institute, Inc., Cleveland, Ohio, 1963.
5. Schlichting, Hermann: Boundary Layer Theory. Fourth ed., McGraw-Hill Book Co., Inc., 1962, pp. 53-54.
6. Constantinescu, V. N.: Theory of Turbulent Lubrication. Lubrication and Wear. D. Muster and B. Sternlicht, eds., McCutchan Pub. Corp., 1965, pp. 153-213.

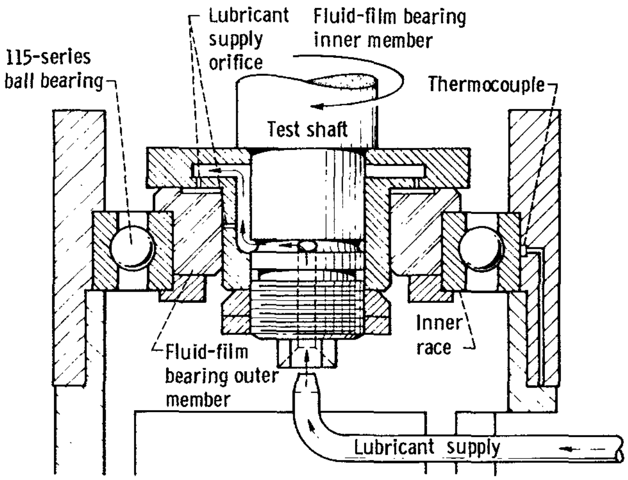
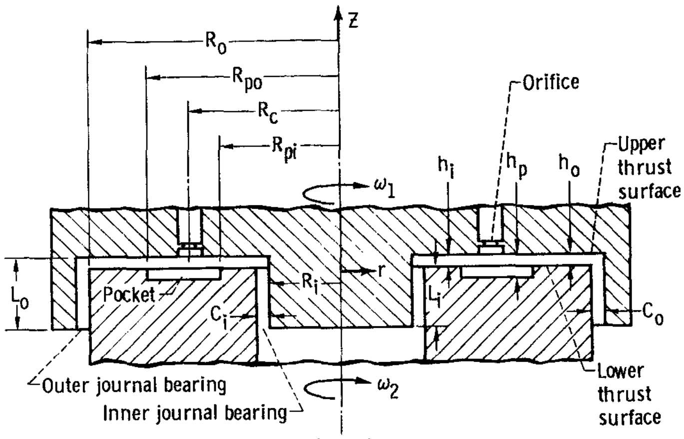
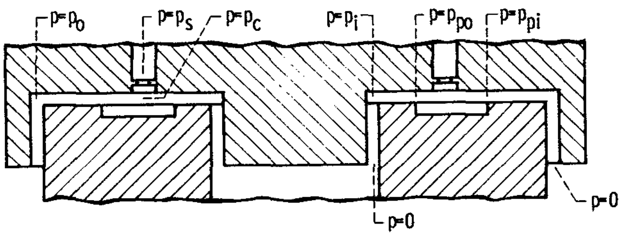


Figure 1. - Series-hybrid rolling-element bearing including pressurized thrust bearing.



(a) Dimensions.



(b) Pressures.

Figure 2. - Pressurized thrust bearing.

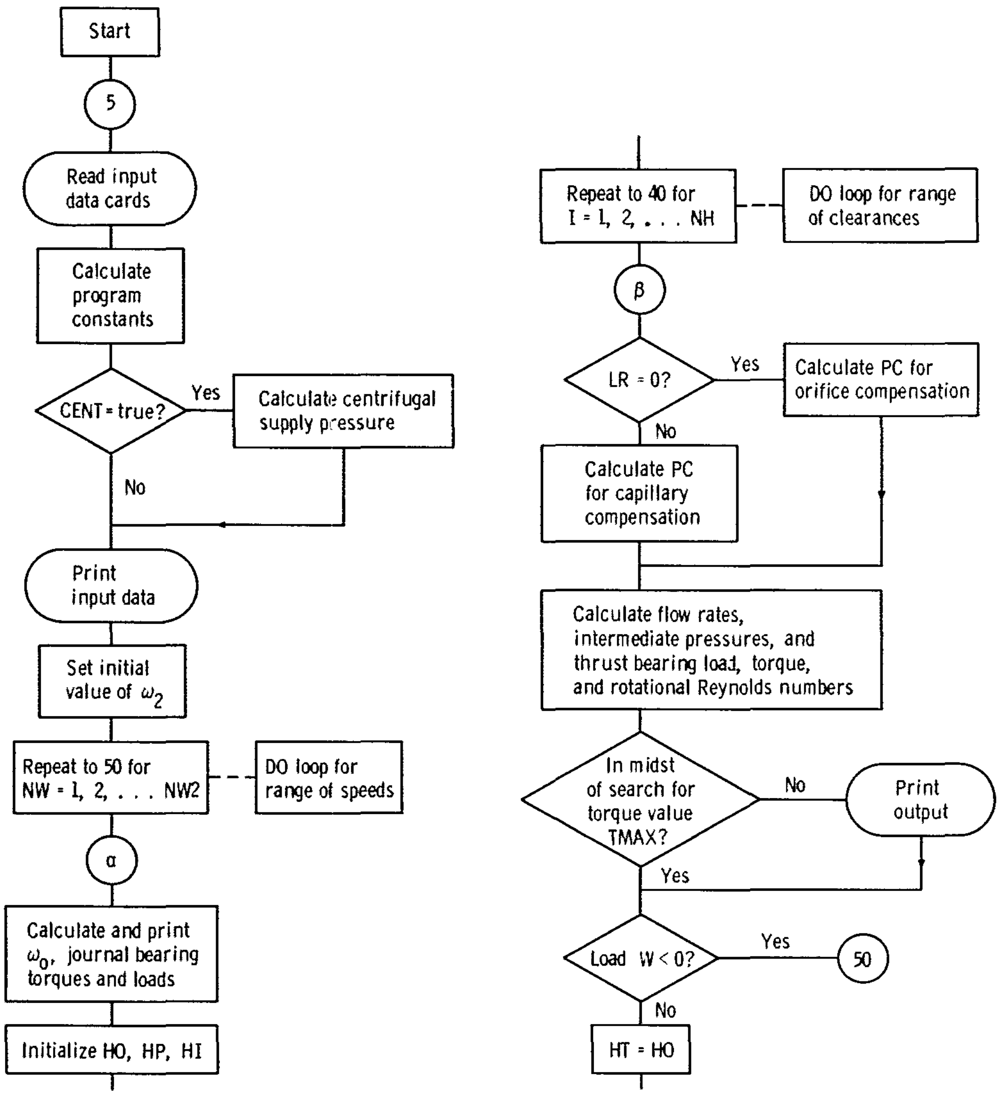


Figure 3. - Flow chart of program.

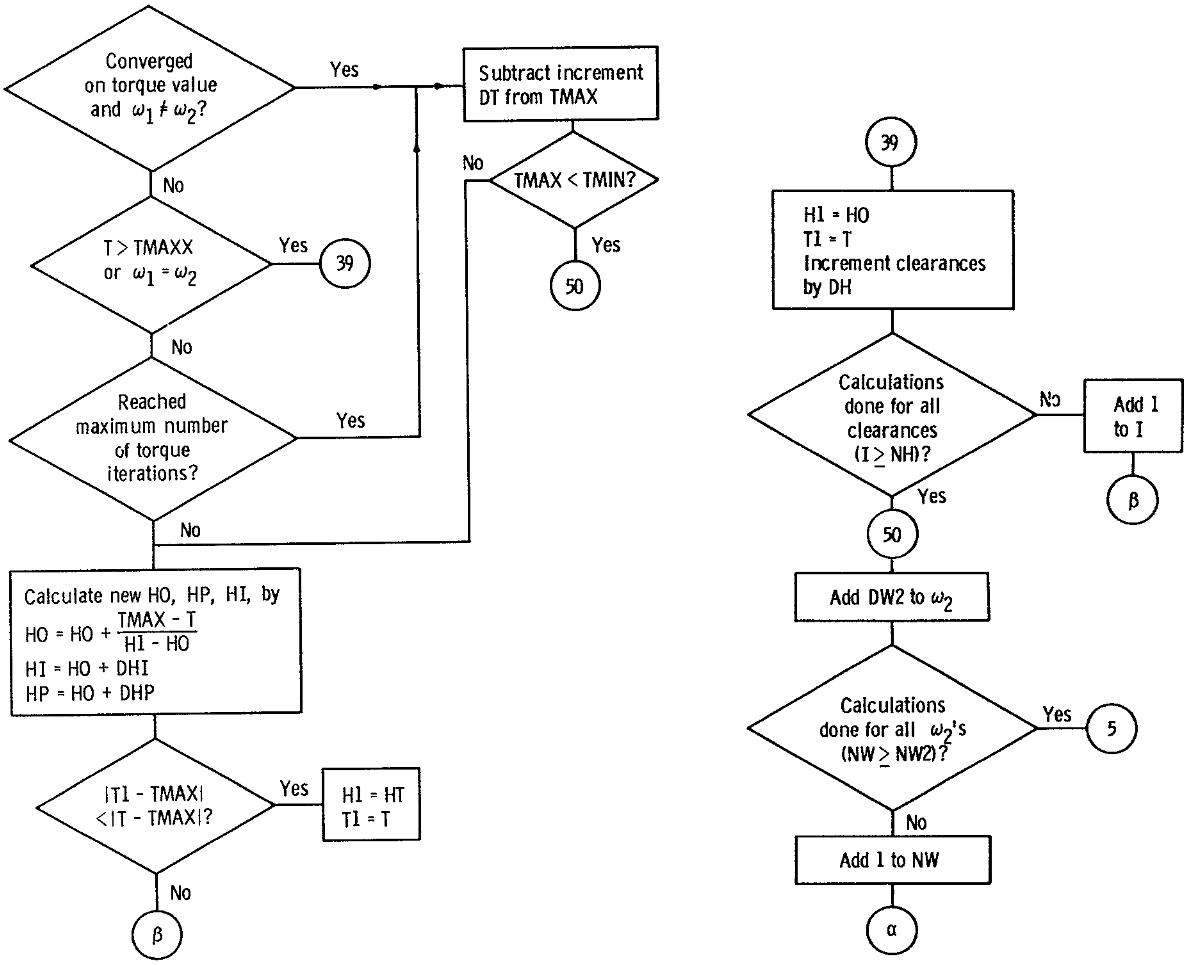


Figure 3. - Concluded.

TITLE	PROJECT NUMBER	ANALYST	SHEET ____ OF ____

STATEMENT NUMBER | CONT | FORTRAN STATEMENT | IDENTIFICATION

```
 $BRG CD=.6,CI=.001,D=.009,DH=.0001,DW2=6000.,HII=.04,HOO=0.,HPP=.009,LI=.86,
 LO=0.,LR=0.,MU=9.4E-7,N=4,NH=16,NW2=3,RC=1.2,RHO=60.,RI=.875,RO=1.6,RPI=.925,
 RPO=1.375,TMAXX=0.,W1=12000.,W22=0.,CENT=.TRUE.,METRIC=.FALSE.$
```

NASA-C-836 (REV 9-14-59)

CD-6729

Figure 4. - Example of input data.

```
  ANALYSIS OF ORIFICE COMPENSATED THRUST BEARING

$BRG

CD     =   6.0000000E-01, CI     =   1.0000000E-03, CO     =   1.0000000E 00, D      =   8.9999999E-03, DH     =   1.0000000E-04,

DT     =   C.            , DW2    =   6.0000000E 03, HII    =   4.0000000E-02, HOO    =   0.            , HPP    =   8.9999999E-03,

LI     =   8.5999999E-01, LO     =   0.            , LR     =   0.            , MU     =   9.4000000E-07, N      =               4,

NH     =              16, NW2    =               3, PS     =   1.0227569E 02, RC     =   1.2000000E 00, RHO    =   5.0000000E 01,

RHUM   =   8.9953942E-05, RI     =   8.7500000E-01, RO     =   1.6000000E 00, RPI    =   9.2499999E-01, RPO    =   1.3750000E 00,

TMAXX  =   C.            , TMIN   =   0.            , W1     =   1.2000000E 04, W22    =   0.            ,

CENT   =        T, METRIC=       F,

$ END

  W1    12000.0 RPM,    CENTRIFUGAL SUPPLY PRESSURE 102.3    PSI AT RC = 1.200    INCHES


  W2        0.  RPM,    W1   12000.0 RPM,    W0    6572.7 RPM
  JOURNAL BEARING TORQUES TJO,TJI, IN LB  0          4.276       LOADS AT ECC RATIO = 0.50  WJO,WJI, LB  0          493.3
   1         2          3          4          5          6           7          8           9          10         11
HO,HI,HP  LOAD,LB  FLOW,IN3/SEC  WO,WI,LB   WPO,WPI,LB  TO,TI,TP    QO,QI      PC,PPI,PI    PPO,PO  REYNOLDS NR R(PMIN),PMIN
  MILS    TORQUE,IN LB                                 IN LB      IN3/SEC       PSI         PSI    REO,REI,REP   DPDR(RO)

  C.      444.35      0.474E-01   106.6       145.5      0.100E 07  0            97.95       107.6      0            4.343
    40.00      0.100E 07              23.89       163.4      0.677E-02  -0.474E-01   85.47       0.107E-05  0.445E 04  -455.3
     9.00                                                    0.586                   83.55                  0.149E 04  -434.2

  C.100  444.01       0.478E-01   106.6       145.4      55.28      0.447E-03    97.88       107.5      19.24        4.342
    40.10      60.14                  23.87       168.2      0.675E-02  -0.473E-01   85.39       0.107E-05  0.446E 04  -454.8
     9.10                                                    0.580                   83.48                  0.150E 04  -433.9

  C.200  441.57       0.506E-01   106.0       144.6      27.64      0.356E-02    97.34       106.9      38.48        4.332
    40.20      32.49                  23.72       167.3      0.673E-02  -0.470E-01   84.86       0.107E-05  0.447E 04  -451.3
     9.20                                                    0.573                   82.94                  0.152E 04  -431.7

  C.300  434.46       0.580E-01   104.5       142.3      18.43      0.119E-01    95.79       105.4      57.72        4.304
    40.30      23.28                  23.28       164.4      0.672E-02  -0.461E-01   83.31       0.107E-05  0.448E 04  -441.1
     9.30                                                    0.567                   81.39                  0.154E 04  -425.3

  C.400  419.00       0.716E-01   101.1       137.4      13.82      0.273E-01    92.40       102.0      76.96        4.243
    40.40      18.66                  22.33       158.2      0.670E-02  -0.442E-01   79.92       0.107E-05  0.449E 04  -419.1
     9.40                                                    0.561                   78.00                  0.155E 04  -411.3

  C.500  390.79       0.913E-01   94.90       128.5      11.06      0.506E-01    86.20       95.81      96.20        4.128
    40.50      15.89                  20.58       146.8      0.668E-02  -0.407E-01   73.73       0.107E-05  0.451E 04  -379.8
     9.50                                                    0.555                   71.81                  0.157E 04  -385.7

  C.600  347.52       0.115       85.40       114.9      9.213      0.798E-01    76.69       86.30      115.4        3.946
    40.60      14.04                  17.89       129.4      0.667E-02  -0.353E-01   64.22       0.107E-05  0.452E 04  -321.7
     9.60                                                    0.549                   62.31                  0.159E 04  -346.5

  C.700  292.23       0.140       73.26       97.48      7.897      0.111        64.53       74.14      134.7        3.700
    40.70      12.72                  14.45       107.0      0.665E-02  -0.284E-01   52.07       0.107E-05  0.453E 04  -251.8
     9.70                                                    0.544                   50.15                  0.160E 04  -296.4
```

Figure 5. - Example of computer program output.

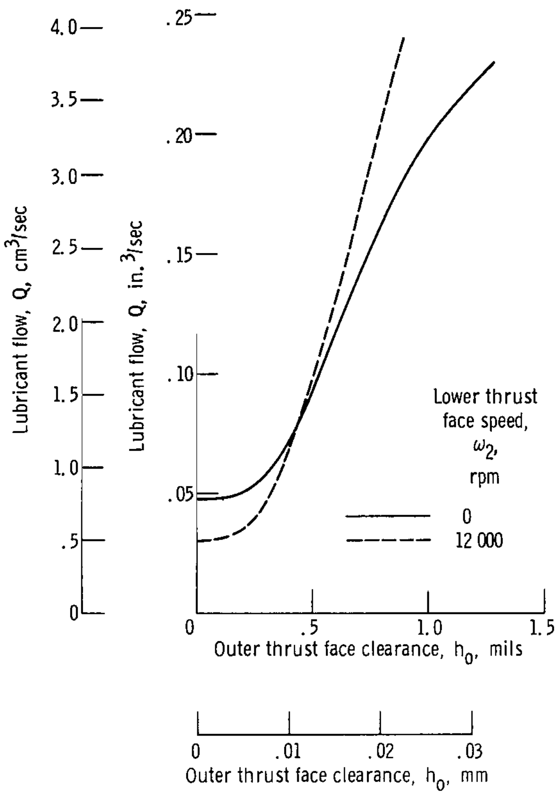


Figure 7. - Variation of lubricant flow with clearance. Upper thrust face speed, 12 000 rpm.

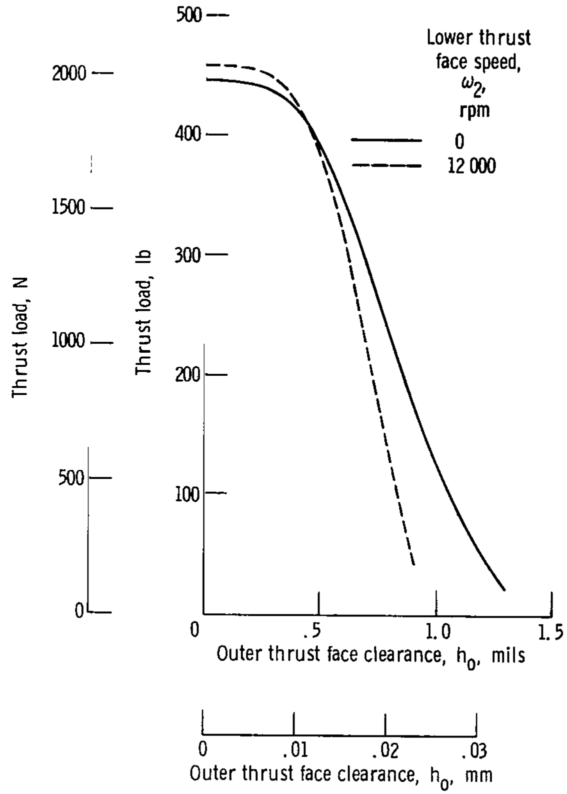


Figure 6. - Variation of load with clearance. Upper thrust face speed, 12 000 rpm.

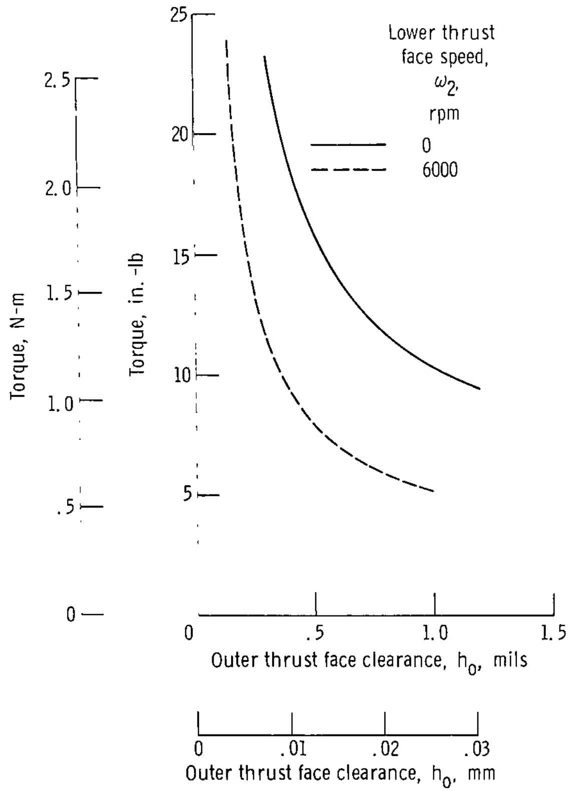


Figure 8. - Torque of bearing assembly. Upper thrust face speed, 12 000 rpm.

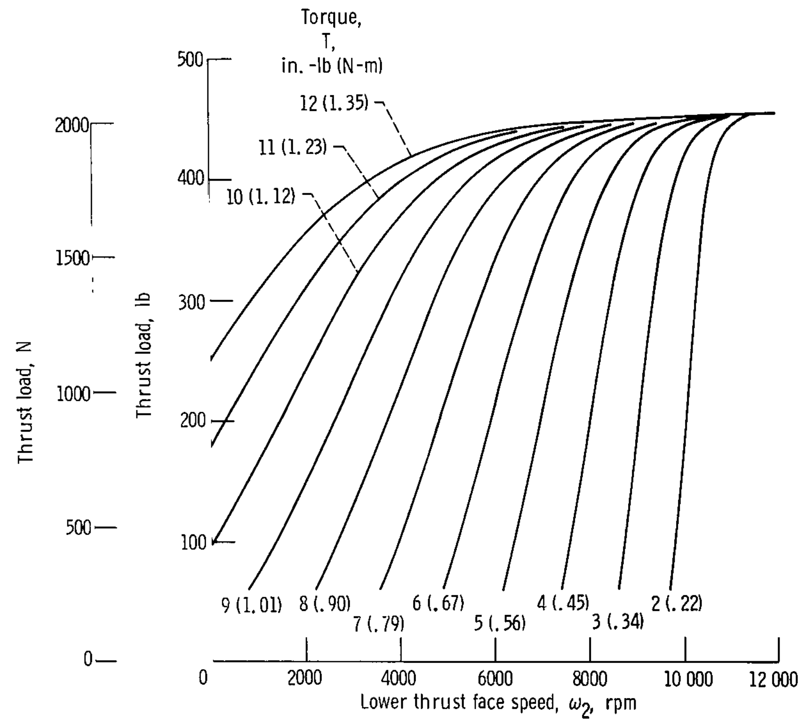


Figure 9. - Thrust load as function of torque and lower thrust face speed. Upper thrust face speed, 12 000 rpm.

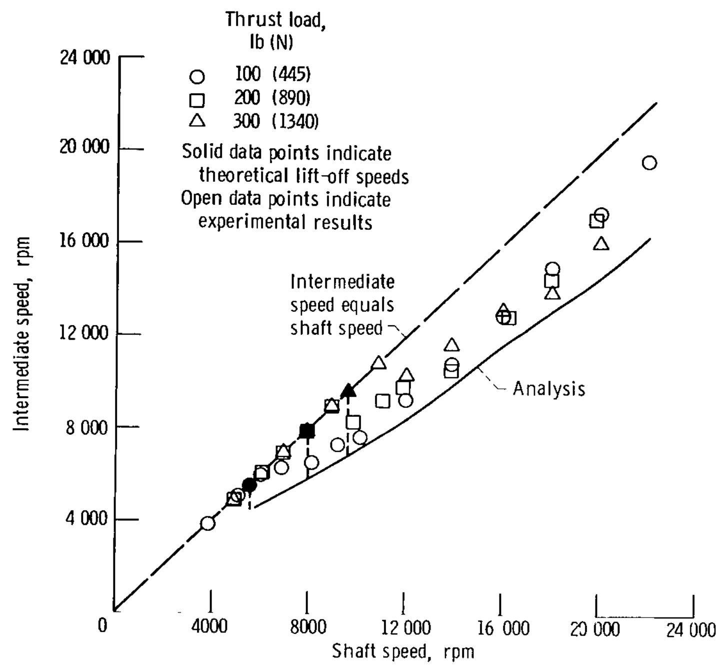


Figure 10. - Comparison of theoretical and experimental speed sharing of the series hybrid bearing.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
WASHINGTON, D. C. 20546

OFFICIAL BUSINESS

FIRST CLASS MAIL

POSTAGE AND FEES PAID
NATIONAL AERONAUTICS AND
SPACE ADMINISTRATION

03U 001 40 51 3DS 71028 00903
AIR FORCE WEAPONS LABORATORY /WLOL/
KIRTLAND AFB, NEW MEXICO 87117

ATT E. LOU BOWMAN, CHIEF,TECH. LIBRARY

POSTMASTER: If Undeliverable (Section 158 Postal Manual) Do Not Return

"The aeronautical and space activities of the United States shall be conducted so as to contribute . . . to the expansion of human knowledge of phenomena in the atmosphere and space. The Administration shall provide for the widest practicable and appropriate dissemination of information concerning its activities and the results thereof."

—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

NASA SCIENTIFIC AND TECHNICAL PUBLICATIONS

TECHNICAL REPORTS: Scientific and technical information considered important, complete, and a lasting contribution to existing knowledge.

TECHNICAL NOTES: Information less broad in scope but nevertheless of importance as a contribution to existing knowledge.

TECHNICAL MEMORANDUMS: Information receiving limited distribution because of preliminary data, security classification, or other reasons.

CONTRACTOR REPORTS: Scientific and technical information generated under a NASA contract or grant and considered an important contribution to existing knowledge.

TECHNICAL TRANSLATIONS: Information published in a foreign language considered to merit NASA distribution in English.

SPECIAL PUBLICATIONS: Information derived from or of value to NASA activities. Publications include conference proceedings, monographs, data compilations, handbooks, sourcebooks, and special bibliographies.

TECHNOLOGY UTILIZATION PUBLICATIONS: Information on technology used by NASA that may be of particular interest in commercial and other non-aerospace applications. Publications include Tech Briefs, Technology Utilization Reports and Technology Surveys.

Details on the availability of these publications may be obtained from:

SCIENTIFIC AND TECHNICAL INFORMATION OFFICE

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Washington, D.C. 20546